International Journal of Heat and Mass Transfer 94 (2016) 403–410

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Experimental analysis of steam condensation in vertical tube with small diameter

Milan Kubín, Jiří Hirš, Josef Plášek *

Brno University of Technology, Faculty of Civil Engineering, Veveří 331/95, Brno 602 00, Czech Republic

ARTICLE INFO

Article history:
Received 14 September 2015
Received in revised form 9 November 2015
Accepted 9 November 2015
Available online 11 December 2015

Keywords:
Water steam
Condensation
Vertical tube
Heat transfer coefficient

ABSTRACT

Thermal design of tubular heat exchanger based on condensation heat of water steam requires knowledge of condensation heat transfer in the each tube. This paper is just aimed on experimental analysis of steam condensation in vertical copper tube in length of 1285 mm with 2.0 mm inner diameter and 0.5 mm wall thickness. Experimental measurement is performed in 12 steps with variable inlet temperature and mass flow rate of water steam. The heat transfer coefficient on the inner surface of tube in condensation zone is calculated by Thermal resistance method and Wilson plot method. The correlation quality of results obtained from both methods is 98.8%. The results are compared with other experimental studies and also correlated with five chosen equations for prediction of condensation heat transfer coefficient. The correlation quality of results obtained from this experimental analysis and four tested equations is over 96.6%, only theoretically determined Nusselt equation undervalues condensation heat transfer coefficient as is known. The Nusselt equation does not take to account waves on condensate surface. These waves on condensate surface are caused by flow of water steam in tube and the wave's effect is approximately 20.5% in this presented case.

1. Introduction

Heat exchangers are commonly part of technology systems and tubular heat exchanger is one of often applied types. The most efficient tubular heat exchangers use latent heat of fluids as is phase change from liquid to gas (evaporation) or reverse phase change from gas to liquid (condensation). Thermal design of tubular heat exchanger based on condensation heat requires knowledge of phase change process in the tubes. This paper is focused on experimental analysis of steam condensation in vertical tube with small diameter.

The first article about laminar film condensation is published by Nusselt [1] in 1916, where Nusselt analytically expressed condensation heat transfer coefficient dependent on amount of steam condensate. The Nusselt equation (see Eq. (11)) assumes smooth and uniform liquid film on wall surface and condensation heat transfer coefficient is equal to ratio of thermal conductivity and thickness of film condensate. The effect of sub-cooling condensate on surface wall is published later by Bromley [2] and non-linear temperature distribution in film condensate is studied by Rohsenow [3]. The classical Nusselt theory is also extended regard to momentum changes of film condensate by Sparrow [4–6] and stability of laminar flow down of film condensate is published by Bankoff [7] or Marschall and Lee [8] and recently others [9–11].

The Nusselt equation with assumption of smooth liquid film on wall surface is valid for stationary steam, because the flowing steam in tube causes waves on condensate surface and these waves improve condensation heat transfer. The wave's effect is studied by Kapitsa [12] in 1948 and later McAdams [13] recommended multiplied Nusselt equation by the wave's effect factor 1.2 as a discrepancy correction between experimental results and theoretical Nusselt solution. The Nusselt equation is increased about 20.6% by Whitham [14] as the wave's effect on condensation heat transfer coefficient, see Eq. (12). The next theoretically determined equation (see Eq. (13)) which includes the wave's effect is published by Hobler [15] and the wave's effect is continuously studied for example in [16–19]. The equation for prediction of condensation heat transfer coefficient can be also determined by experimental way and usually is formulated in exponential function. The bases of exponential function are often fluid properties (*Nu*, *Pr*, *Re* etc.) for example Hausen [20] in Eq. (15) or boundary conditions (q, p, ΔT etc.) where the exponents of bases are determined by experimental measurement, for example Kutateladze [21] in Eq. (14).

* Corresponding author.
E-mail address: plasek.j@fce.vutbr.cz (J. Plášek).

http://dx.doi.org/10.1016/j.ijheatmasstransfer.2015.11.022
0017-9310/

Nomenclature

Latin symbols

A	slope parameter of linear regression function [m^{-1}]
B	constant term of linear regression function [m K W^{-1}]
c	specific thermal capacity [J kg^{-1} K^{-1}]
C	multiple constant in exponential function [–]
d	characteristic length in Nusselt number [m]
D	diameter of tube [m]
g	gravity acceleration [m s^{-2}]
h	specific enthalpy [J kg^{-1}]
H	level of steam condensate [m]
k	overall heat transfer coefficient [W m^{-1} K^{-1}]
l_{23}	latent heat of phase change [J kg^{-1}]
L	total length of tube [m]
m	mass flow rate [kg s^{-1}]
n	total count of tubes [pcs]
p	static pressure [Pa]
q	specific heat flux [W m^{-2}]
Q	total heat flux [W]
R	Thermal resistance [m K W^{-1}]
t	temperature in Celsius scale [°C]
T	temperature in Kelvin scale [K]
ΔT	logarithmic mean temperature difference [K]
V	volume flow rate [m^3 s^{-1}]
x	variable on x-axis [–]
y	variable on y-axis [–]

Greek symbols

α	heat transfer coefficient [W m^{-2} K^{-1}]
ε	percentage differences [%]
λ	thermal conductivity [W m^{-1} K^{-1}]
μ	dynamic viscosity [Pa s]
ν	kinematic viscosity [m^2 s^{-1}]
π	mathematical constant [–]
ρ	bulk density [kg m^{-3}]
σ	surface tension [N m^{-1}]

Subscripts

c	condensate
e	external
i	internal
min	minimal value
T	wall of tube
v	vapour
w	water
in	inlet
out	outlet

Dimensionless numbers

Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number

The theme of water steam condensation is continuously studied in many articles by theoretical or experimental way, for example [22–25] and recently others [26–29]. Water steam condensation in vertical tube is also recently studied in article [30], but the study is for wide tube with ratio of length to inner diameter $L/D = 1.10$. The purpose of this paper is parametric experimental study of vertical copper tube with ratio of length to inner diameter $L/D = 642.50$ [–]. Impact of variable inlet temperature and mass flow rate of water steam on condensation heat transfer coefficient is also studied. The obtained results are correlated with five chosen equations for prediction of condensation heat transfer coefficient and compared with other experimental studies [25,31–34]. The wave's effect on condensation heat transfer coefficient is evaluated, too.

2. Experimental setup

The experimental analysis is realized in 12 steps on vertically oriented copper tube in length of 1285 mm with 2.0 mm inner diameter and 0.5 mm wall thickness. Tubes are intentionally measured in bundle of 37 tubes, because some small geometric imperfections of each tube are eliminated. Concurrently measured values in the bundle more correspond with statistical average and edge effect is eliminated, too. The bundle of 37 tubes is surrounded by outer copper tube of diameter 400 mm. The experimental setup can be for description divided to loop of water steam and loop of cooling water, see Fig. 1.

The loop of water steam is composed from steam generator (A), where water steam is produced with known temperature $t_{v,in}$ [°C] and pressure $p_{v,in}$ [Pa]. After that water steam enters into measured bundle of 37 copper tubes (B), where the condensation process is realized. Volume flow rate of condensate V_c [m^3/s] and temperature of condensate $t_{c,out}$ [°C] is measured on outflow from the bundle before collection tank (C). The interspace of bundle is counter-flow cooled by loop from source of cold water (D). Inlet temperature $t_{w,in}$ [°C] and volume flow rate V_w [m^3/s] of cooling water is monitored on enter to the bundle of tubes. Outlet temperature $t_{w,out}$ [°C] and pressure $p_{w,out}$ [Pa] of cooling water is measured on outflow from the bundle of tubes. The volume changes of cooling water are compensated by expansion vessel (E). The steam condensate level H [m] in the measured tubes is displayed on external gauge level (G).

The inlet temperature of water steam is changed in 12 steps during the experimental measurement in range from $t_{v,in} = 100.2$ °C to $t_{v,in} = 117.9$ °C. Concurrently is changed mass flow rate of water steam in range from $m_v = 0.00898$ kg/s to $m_v = 0.01154$ kg/s. The changed input parameters of water steam are kept for a sufficiently long period to obtain of thermal steady state, according to monitored values. Mass flow rate $m_w = 0.274 \pm 0.001$ kg/s and inlet temperature $t_{w,in} = 11.0 \pm 0.2$ °C of cooling water is almost on constant value for the whole time, more Table 1. The uncertainty of certificated gauges is on temperature sensor ±0.3%, pressure gauge ±0.6%, external gauge level ±0.2% and uncertainty of volume flow rate is ±0.5%. The internal surface of measured tubes is cleaned by high percentage alcohol cleaner and the purity of water steam from steam generator is over 99.9%

3. Solution methods

The transferred condensation heat Q_v [W] between water steam and cooling water is calculated from Eq. (1), where specific enthalpy of water steam condensate is $h_{c,out} = 419.10$ kJ/kg and condensation temperature is $t_{v,out} = 100$ °C. The logarithmic mean temperature difference ΔT [K] for counter-flow involvement is determined from Eq. (2). The one-dimensional state steady overall heat transfer coefficient k [W/(m K)] for cylindrical wall in

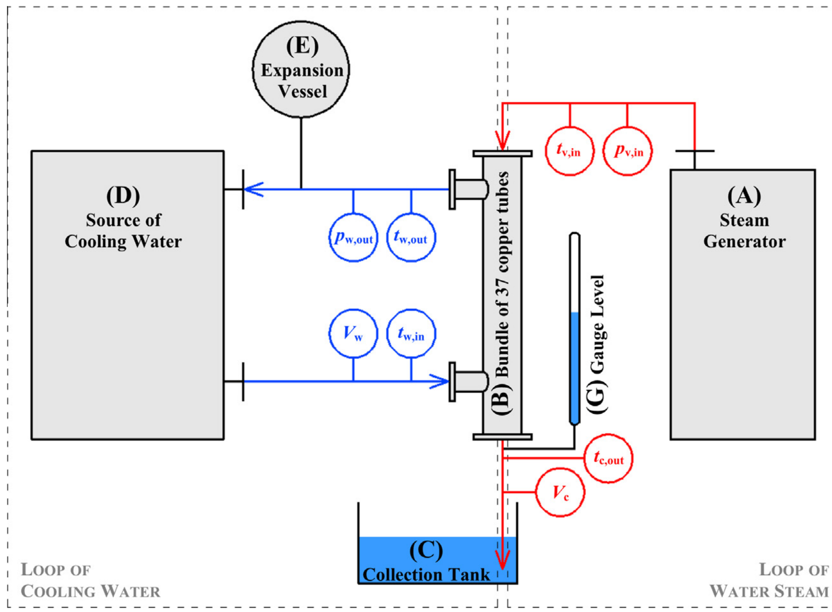

Fig. 1. Scheme of experimental setup.

Table 1
Experimentally measured values in 12 steps.

Case [No]	Q_v [W]	q_v [W/m^2]	k [W/(m K)]	H [m]	Parameters of water steam				Parameters of cooling water		
					m_v [kg/s]	$p_{v,in}$ [kPa]	$t_{v,in}$ [°C]	$t_{c,out}$ [°C]	m_w [kg/s]	$t_{w,in}$ [°C]	$t_{w,out}$ [°C]
1.	20,266	73,494	9.326	0.385	0.00898	102.2	100.2	34.0	0.2729	10.8	30.7
2.	20,302	75,382	9.366	0.399	0.00901	112.8	103.0	33.7	0.2745	10.8	30.6
3.	20,598	81,370	9.484	0.435	0.00905	162.7	113.8	34.5	0.2732	10.9	31.1
4.	21,105	83,412	9.521	0.435	0.00925	185.8	117.9	34.5	0.2737	10.9	31.5
5.	21,916	73,241	9.321	0.332	0.00970	108.9	102.0	47.0	0.2743	11.2	32.1
6.	22,843	81,329	9.483	0.373	0.01002	177.6	116.5	47.8	0.2751	11.1	32.8
7.	23,123	81,296	9.483	0.365	0.01014	180.5	117.0	47.8	0.2735	11.1	33.1
8.	24,078	78,449	9.429	0.315	0.01058	161.6	113.6	50.1	0.2732	11.2	34.1
9.	24,252	78,171	9.422	0.308	0.01066	159.0	113.1	49.4	0.2745	11.1	34.1
10.	25,527	70,998	9.270	0.198	0.01129	113.2	103.1	50.9	0.2743	11.1	35.3
11.	25,858	74,578	9.349	0.226	0.01139	141.5	109.6	39.4	0.2735	10.8	35.9
12.	26,161	72,871	9.313	0.199	0.01154	130.9	107.3	39.9	0.2743	10.9	36.2
Uncertainty	±1.1%	±1.1%	±1.8%	±0.2%	±0.5%	±0.6%	±0.3%	±0.3%	±0.5%	±0.3%	±0.3%

condensing zone is calculated from Eq. (3). Finally the condensation heat transfer coefficient α_v [W/(m^2 K)] is determined by Thermal resistance method and Wilson plot method.

$$Q_v = m_v \cdot (h_{v,in} - h_{c,out}) \tag{1}$$

$$\Delta T = \frac{(t_{v,in} - t_{w,out}) - (t_{v,out} - t_{w,in})}{\ln\left(\frac{t_{v,in} - t_{w,out}}{t_{v,out} - t_{w,in}}\right)} \tag{2}$$

$$k = \frac{Q_v}{n_T \cdot (L - H) \cdot \Delta T} \tag{3}$$

3.1. Thermal resistance method

The overall heat transfer coefficient k [W/(m K)] of cylindrical wall includes inverted sum of Thermal resistance of solid tube wall R_T [m K W^{-1}] and two unknown surface Thermal resistances on internal R_v [m K W^{-1}] and external surface R_w [m K W^{-1}] of tube, see Eq. (4). The external Thermal resistance R_w [m K W^{-1}] of tube on cooling water site can be estimated by average heat transfer coefficient α_w [W/(m^2 K)] from Eq. (6). The Nusselt number Nu_w [–] in Eq. (6) for cooling water flow along the tube is calculated from Eq. (5), according to Gröber [35]. Subsequently the condensation heat transfer coefficient α_v [W/(m^2 K)] on inner surface of tube can be obtained from Eq. (4).

$$\frac{1}{k} = \frac{1}{\pi \cdot D_i \cdot \alpha_v} + \frac{\ln(D_e/D_i)}{2 \cdot \pi \cdot \lambda_T} + \frac{1}{\pi \cdot D_e \cdot \alpha_w} = R_v + R_T + R_w \tag{4}$$

$$Nu_w = 1.86 \cdot \left(Re_w \cdot Pr_w \cdot \frac{D_e}{L}\right)^{0.33} \tag{5}$$

$$\alpha_w = \frac{Nu_w \cdot \lambda_w}{D_e} \tag{6}$$

3.2. Wilson plot method

The Wilson plot method is suitable for determination of heat transfer coefficient in case where two fluids are separated by solid wall, in detail [36]. The condensation heat transfer coefficient α_v [W/(m^2 K)] on internal surface of tube is expressed by exponential function Eq. (7). The equation Eq. (7) substituted to Eq. (4) is rewritten to linear equation ($y = Ax + B$) where parameter $x = q_v^{-0.80}$ and $y = k^{-1}$, see Eq. (8). The unknown parameters A [m^{-1}] (Eq. (9)) and B [m K W^{-1}] (Eq. (10)) of linear regress function are determined by least square method. The obtained parameters A [m^{-1}] and B [$m\,K\,W^{-1}$] are used for calculation of heat transfer coefficient α_w [$W/(m^2\,K)$] and condensation heat transfer coefficient α_v [$W/(m^2\,K)$].

$$\alpha_v = C \cdot q_v^{0.80} \tag{7}$$

$$\frac{1}{k} = \left(\frac{1}{\pi \cdot D_i \cdot C}\right) \cdot q_v^{-0.80} + \left(\frac{\ln(D_e/D_i)}{2 \cdot \pi \cdot \lambda_T} + \frac{1}{\pi \cdot D_e \cdot \alpha_w}\right) \tag{8}$$

$$A = \frac{1}{\pi \cdot D_i \cdot C} \tag{9}$$

$$B = \frac{\ln(D_e/D_i)}{2 \cdot \pi \cdot \lambda_T} + \frac{1}{\pi \cdot D_e \cdot \alpha_w} \tag{10}$$

3.3. Predicted condensation heat transfer coefficient

The condensation heat transfer coefficient α_v [$W/(m^2\,K)$] can be predicted by equations obtained by theoretical or experimental way. The first chosen equation is theoretically determined by Nusselt [1] in 1916. The Nusselt equation (Eq. (11)) is expressed from amount of condensate and Thermal resistance of laminar film condensate on surface wall.

$$\alpha_v = 0.9428 \cdot \left[\frac{g \cdot \rho_c \cdot l_{23} \cdot \lambda_c^3}{\nu_c \cdot (t_v - t_T) \cdot L}\right]^{0.25} \tag{11}$$

The Nusselt equation (Eq. (11)) is valid for stationary steam because flowing steam in tube causes waves on condensate surface. The wave's effect increases condensation heat transfer about 20.6% as published Whitham [14] in Eq. (12).

$$\alpha_v = 1.137 \cdot \left[\frac{g \cdot \rho_c \cdot l_{23} \cdot \lambda_c^3}{\nu_c \cdot (t_v - t_T) \cdot L}\right]^{0.25} \tag{12}$$

Next chosen equation (Eq. (13)) is theoretically determined for calculation of condensation heat transfer coefficient and includes the wave's effect, too. This equation is chosen for comparison because the equation is often applied in engineering tasks. The equation (Eq. (13)) published by Hobler [15] is valid for many kind of fluids with pressure $0.07 < p_v$ [MPa]< 17 and specific heat flux $1.0 < q_v$ [kW/m^2]< 1000.

$$\alpha_v = 0.00252 \cdot \left(\frac{\rho_v \cdot l_{23}}{\rho_c - \rho_v} \cdot \frac{\rho_c}{\sigma_c}\right)^{0.33} \cdot \frac{\lambda_c^{0.8} \cdot q_v^{0.7}}{\mu_c^{0.5} \cdot c_c^{0.167} \cdot T_v^{0.37}} \cdot p_v^{\frac{10}{T_v - 273.15}} \tag{13}$$

Another chosen equation (Eq. (14)) determined by experimental way is formulated in typical exponential function $\alpha = C \cdot q^n$ similar as substitution in Wilson plot method, see Eq. (7). The base of function is specific heat flux q [W/m^2] and prefix constant $C = 1.537$ depends on kind of surface and fluid properties, more Kutateladze [21]. The exponent of function takes into account boundary conditions and for constant boil temperature without impact of radiation heat transfer is $n = 0.75$.

$$\alpha_v = 1.537 \cdot q_v^{0.75} \tag{14}$$

The last chosen equation (Eq. (15)) for comparison is determined by experimental way and predict minimal value of Nusselt number Nu_{min} [–] depending on fluid properties included in Prandtl number Pr_c [–]. The characteristics length d [m] in Nusselt number Nu_{min} [–] is $d = (0.125 \cdot \nu_c{}^2)^{0.33}$, according to Hausen [20].

$$Nu_{\min} = 0.16 \cdot Pr_c^{0.61} \tag{15}$$

4. Results and discussion

The experimental measurement is performed on bundle of 37 vertical tubes in length of 1285 mm with inner diameter 2.0 mm and 0.5 mm wall thickness. The ratio of tube length to inner diameter is $L/D = 642.50$ [–]. The inlet temperature $t_{v,in}$ [°C] and mass flow rate m_v [kg/s] of water steam is changed in 12 steps during the experimental measurement. The mass flow rate m_w [kg/s] and inlet temperature $t_{w,in}$ [°C] of cooling water is kept almost on constant values, see Table 1. The overall heat transfer coefficient $k = 9.397 \pm 0.125$ W/(m K) with accuracy ±1.8% is obtained from Eq. (3) and plotted in Fig. 2. The condensation heat transfer coefficient α_v [$W/(m^2\,K)$] is determined by Thermal resistance method and Wilson plot method.

4.1. Results from Thermal resistance method

The condensation heat transfer coefficient $\alpha_v = 7139 \pm 468$ $W/(m^2\,K)$ with accuracy ±6.6% is calculated from overall heat transfer coefficient k [W/m K] with use Eq. (4) and compared with Wilson plot method in Fig. 3. The correlation quality of condensation heat transfer coefficient obtained from Thermal resistance method and Wilson plot method is 98.8%. The heat transfer coefficient $\alpha_w = 1264 \pm 1.8$ $W/(m^2\,K)$ on external surface of tube (site of cooling water) is predicted by Gröber [35] in Eq. (5). Thermal resistance method solves each measured step separately therefore inaccuracies are not eliminated with other measured steps.

4.2. Results from Wilson plot method

Wilson plot method is more reliable than Thermal resistance method, because the Wilson plot method is based on trend observation and inaccuracies are mutually eliminated by linear regress function of least squares method, see Fig. 4. The condensation heat transfer coefficient is $\alpha_v = 7229 \pm 475$ $W/(m^2\,K)$ with accuracy ±6.6% obtained by linear regress function (Eq. (8)). The slope parameter in linear regress function is $A = 178.685$ [m^{-1}], constant term parameter $B = 0.08437$ [$m\,K\,W^{-1}$] and constant $C = 0.8907$ [–] in Eq. (7). The correlation quality of experimentally measured values and linear regression function is 99%. The heat transfer coefficient on external surface of tube (site of cooling water) is determined on $\alpha_w = 1260.1$ $W/(m^2\,K)$ by Eq. (10). The constant term parameter B [$m\,K\,W^{-1}$] includes Thermal resistance of copper tube wall $R_T = 0.1635 \cdot 10^{-3}$ m K/W and surface Thermal resistance of cooling water on value $R_w = 84.20 \cdot 10^{-3}$ m K/W.

4.3. Comparison of results with predicted condensation heat transfer coefficient

The first chosen equation (Eq. (11)) for prediction of condensation heat transfer coefficient is theoretically determined by Nusselt [1] and predicts $\alpha_v = 5997 \pm 39$ $W/(m^2\,K)$ with fluctuation ±0.6%. The results obtained from Wilson plot method are about 20.5% higher than condensation heat transfer coefficients predict by Nusselt equation. This fact agrees with studies published by Whitham [14] and Hobler [15]. The second tested equation (Eq. (12)) is published by Whitham [14] and predicts $\alpha_v = 7232 \pm 47$ $W/(m^2\,K)$ with fluctuation ±0.6%. This value is about 1.3% higher than results obtained by Thermal resistance method and also higher about 0.06% than results obtained from Wilson plot method. The third tested equation (Eq. (13)) is published by Hobler [15] and predicts $\alpha_v = 7210 \pm 413$ $W/(m^2\,K)$ with fluctuation ±5.7%. This value is about 0.99% higher than results obtained from Thermal resistance method, but about 0.26% lower than results obtained from Wilson plot method, see Fig. 5.

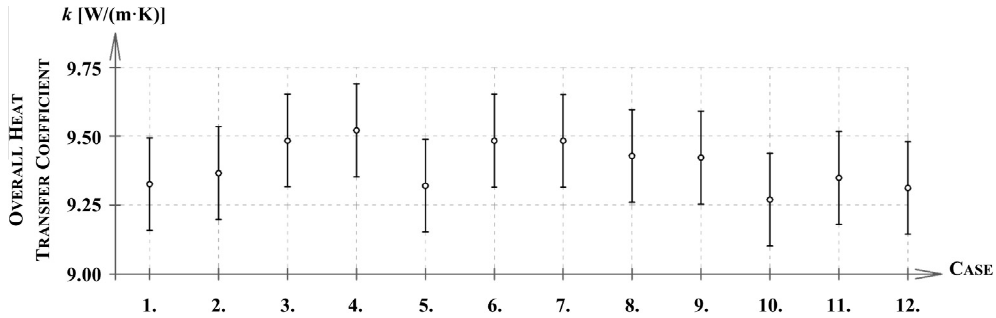

Fig. 2. Overall heat transfer coefficient k [W/m K] determined from experimental measurement.

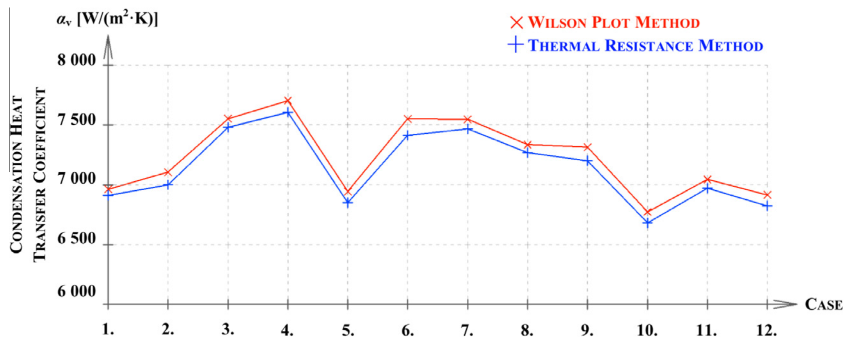

Fig. 3. Comparison of condensation heat transfer coefficient obtained from Thermal resistance method and Wilson plot method.

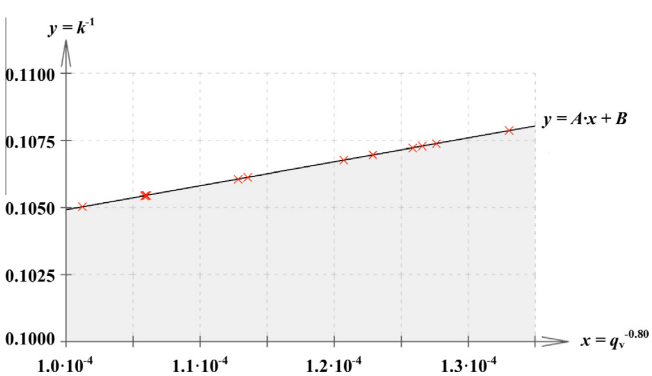

Fig. 4. Linear regress function by least square method.

The fourth tested equation (Eq. (14)) is published by Kutateladze [21] and predicts $\alpha_v = 7106 \pm 437$ W/(m^2 K) with fluctuation ±6.2%. The predicted value of condensation heat transfer coefficient is about 0.46% lower than results obtained from Thermal resistance method and also about 1.6% lower than values obtained by Wilson plot method. Another finding is the Wilson plot method expects exponential shape of substitution function for condensation heat transfer coefficient in Eq. (7) and the chosen equation (Eq. (14)) published by Kutateladze is formulated into same exponential shape $\alpha = C \cdot q^n$. Different is only prefix constant C and exponent n of base. The substitution of exponential function Eq. (7) in the Wilson plot method is correct, because correlation of the equation Eq. (14) with results obtained from Wilson plot method is strong at 98.3%.

The last tested equation (Eq. (15)) published by Hausen [20] is determined by experimental way and predicts minimal value of Nusselt number Nu_{min} [–] according to Prandtl number Pr_c [–] of water steam condensate. In this case is condensation temperature $t_{v,out} = 100$ °C and $Pr_c = 1.75$ for water steam condensate. The Nusselt number is $Nu_{min} = 0.225$ and predicts condensation heat transfer coefficient on value $\alpha_v = 6981.4$ W/(m^2 K). This condensation heat transfer coefficient predicted by equation (Eq. (15)) is lower about 3.4% than results obtained from Wilson plot method and also lower about 2.2% than results obtained from Thermal resistance method. All tested equations for prediction of condensation heat transfer coefficient in Fig. 6 are related to results obtained by Wilson plot method.

4.4. Comparison of results with other experimental studies

The obtained results are comparable with previously published experimental studies about condensation heat transfer coefficient in vertical tube. The first chosen study for comparison published by Al-Shammari et al. [25] is aimed on steam condensation with and without presence of non-condensable gas. The published results of condensation heat transfer coefficient without presence of non-condensable gas are in range $\alpha_v = 4790$ W/(m^2 K) to $\alpha_v = 8518$ W/(m^2 K) with mean value $\alpha_v = 6502$ W/(m^2 K), according to Fig. 8 in study [25]. The second chosen study published by Urban et al. [31] shows condensation heat transfer coefficient in range from $\alpha_v = 4945$ W/(m^2 K) to $\alpha_v = 9191$ W/(m^2 K) with the mean value $\alpha_v = 7285$ W/(m^2 K) in steel tube with inner diameter 6.5 mm. Third chosen study published by Ma et al. [32] is focused on drop-wise and film-wise condensation with presence of non-condensable gas in copper tube with inner diameter 30 mm. The film-wise condensation heat transfer coefficient without presence

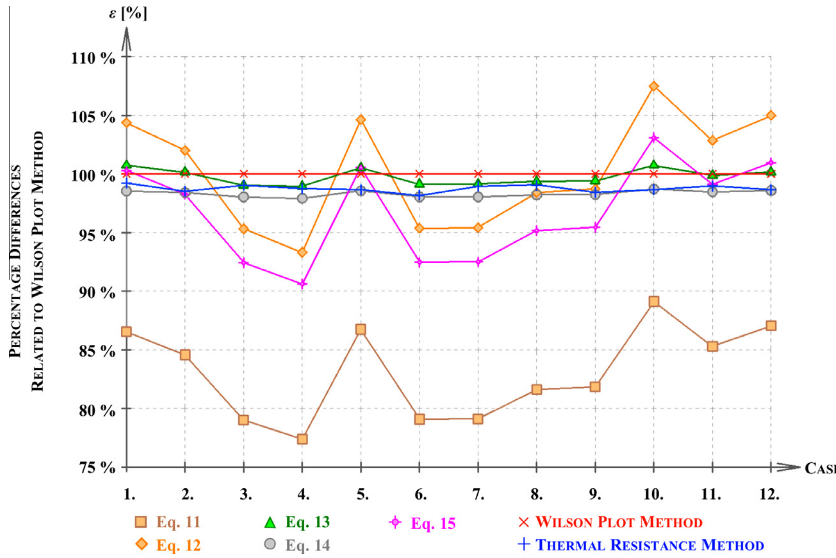

Fig. 5. Correlation of tested equations with results obtained from Wilson plot method.

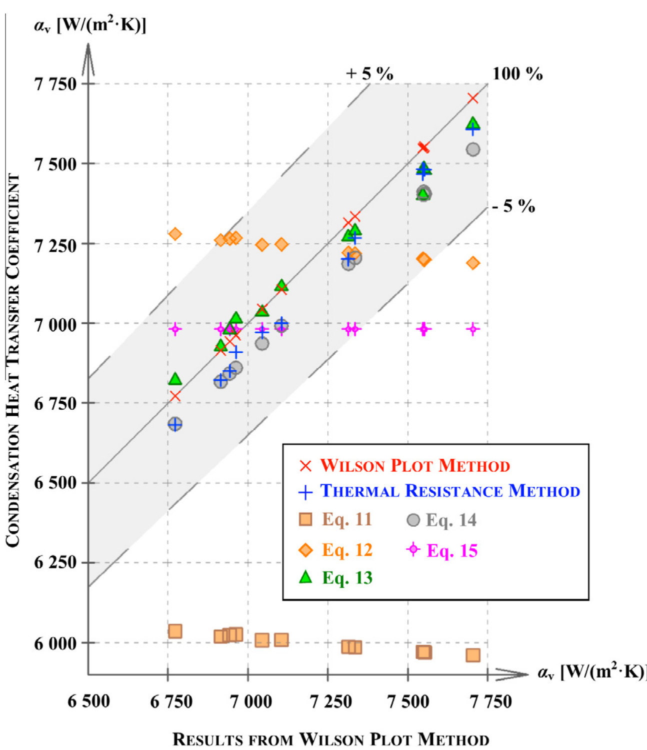

Fig. 6. Comparison of tested equations related to Wilson plot method.

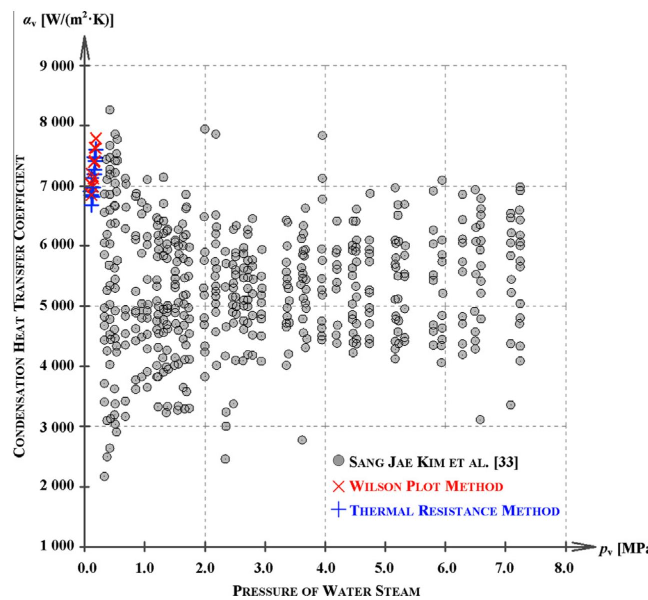

Fig. 7. Correlation of obtained results with study of Kim et al. [33] aimed on film condensation in high pressure steam.

of non-condensable gas is in range from $\alpha_v = 4667\ \mathrm{W/(m^2\,K)}$ to $\alpha_v = 8619\ \mathrm{W/(m^2\,K)}$ with mean value $\alpha_v = 6151\ \mathrm{W/(m^2\,K)}$, according to Fig. 6 in study [32]. The fourth chosen study published by Kim et al. [33] is focused on impact of high pressure steam on condensation heat transfer coefficient. The condensation heat transfer coefficient is in range $\alpha_v = 2170\ \mathrm{W/(m^2\,K)}$ to $\alpha_v = 8270\ \mathrm{W/(m^2\,K)}$ with mean value $\alpha_v = 5443\ \mathrm{W/(m^2\,K)}$ for steam pressure in interval from $p_v = 0.3$ MPa to $p_v = 7.5$ MPa, see in Fig. 7.

The last chosen study for comparison is published by Goodykoontz and Dorsch as a Technical Note of National Aeronautics and Space Administration, see [34]. The document in appendix includes 14 records about experimental measurement of condensation heat transfer coefficient in vertical tube with inner diameter 5/8 inch. The correlation of results is performed in 6 cases with comparable boundary conditions and maximal difference is 23%. All aforementioned experimental studies are summarized in Table 3.

4.5. Influence of mass flow rate

The mass flow rate of water steam is changed during the experimental measurement in range from $m_v = 0.00898$ kg/s to $m_v = 0.01154$ kg/s with increase about 28.5%, see Table 1. The impact of increasing mass flow rate of water steam on condensation heat transfer coefficient is not significant, see Fig. 8.

Table 2
Correlation of obtained results with five tested equations.

Tested equation		α_v [W/(m² K)]	Fluctuation (%)	Correlation (%)
Results from Wilson plot method		7229 ± 475	±6.6	100.0
Results from Thermal resistance method		7139 ± 468	±6.6	98.8
(Eq. (11))	Nusselt equation [1]	5997 ± 39	±0.6	83.0
(Eq. (12))	Whitham equation [14]	7232 ± 47	±0.6	99.9
(Eq. (13))	Hobler equation [15]	7210 ± 413	±5.7	99.7
(Eq. (14))	Kutateladze equation [21]	7106 ± 437	±6.2	98.3
(Eq. (15))	Hausen equation [20]	6981 ± 0	±0.0	96.9

Table 3
Correlation of obtained results with other experimental studies.

Experimental study		Material of tube	D_i [mm]	L [mm]	$p_{v,in}$ [kPa]	$t_{v,in}$ [°C]	α_v [W/(m²·K)]	ε [%]
Results from Wilson plot method		Copper	2.00	1285	⟨102.2; 185.8⟩	⟨100.2; 117.9⟩	7229	100.0
Results from Thermal resistance method		Copper	2.00	1285	⟨102.2; 185.8⟩	⟨100.2; 117.9⟩	7139	98.8
1.	Al-Shammari et al. [25]	Copper	28.25	3000	⟨16.0; 22.0⟩	⟨56.6; 63.18⟩	6502	89.9
2.	Urban et al. [31]	Steel	6.50	1036	226.3	134.9	7285	100.8
3.	Ma et al. [32]	Cooper	30.00	410	100.0	100.0	6151	85.1
4.	Kim and No [33]	Stainless steel	46.20	1800	⟨300; 7500⟩	⟨130; 300⟩	5443	75.3
5.	Goodykoonz and Dorsch [34] (p. 19)	Stainless steel	15.88	2133	111.7	102.2	7008	96.9
6.	Goodykoonz and Dorsch [34] (p. 20)	Stainless Steel	15.88	2133	166.9	114.4	6650	92.0
7.	Goodykoonz and Dorsch [34] (p. 25)	Stainless steel	15.88	2133	266.8	129.4	8455	117.0
8.	Goodykoonz and Dorsch [34] (p. 29)	Stainless steel	15.88	2133	116.5	103.3	8920	123.4
9.	Goodykoonz and Dorsch [34] (p. 31)	Stainless steel	15.88	2133	244.1	126.7	6639	91.8
10.	Goodykoonz and Dorsch [34] (p. 32)	Stainless steel	15.88	2133	243.4	126.7	8160	112.9

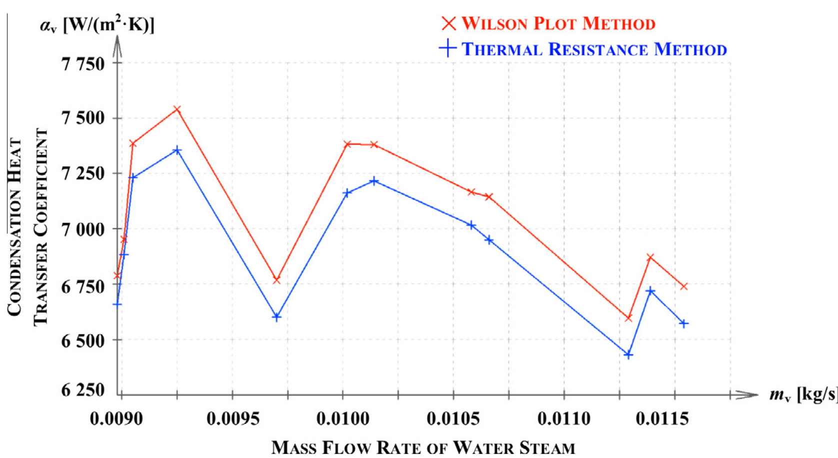

Fig. 8. Insignificant impact of mass flow rate on condensation heat transfer coefficient.

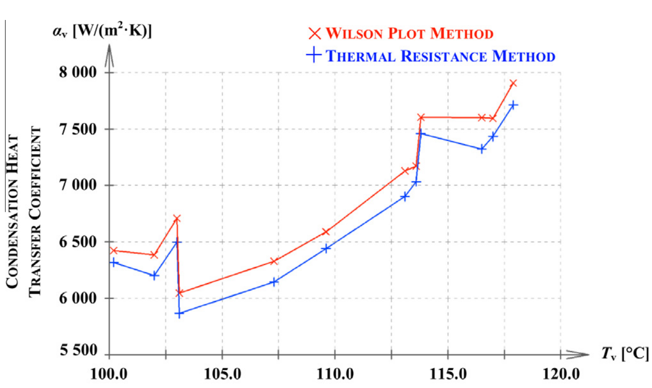

Fig. 9. Insignificant impact of inlet temperature on condensation heat transfer coefficient.

4.6. Influence of inlet temperature

The impact of variable inlet temperature on condensation heat transfer coefficient is experimentally tested in range from $t_{v,in}$ = 100.2 °C to $t_{v,in}$ = 117.9 °C and the inlet steam temperature is increased about 17.6%, see Table 1. The impact of inlet steam temperature on condensation heat transfer coefficient is not significant in this presented case, see Fig. 9.

5. Conclusion

The experimental analysis is performed in 12 steps on vertical copper tube in length of 1285 mm with 2.0 mm inner diameter and 0.5 mm wall thickness. The condensation heat transfer coefficient is calculated by Thermal resistance method and Wilson plot method. The impact of variable mass flow rate and inlet temperature of water steam on condensation heat transfer coefficient is evaluated on parametric experimental measurement. The obtained results are compared with other experimental studies and also correlated with five chosen equations which predict condensation heat transfer coefficient. The main results of this experimental study can be summarized to following points.

(I) The quality correlation of results obtained from Thermal resistance method and Wilson plot method is on value 98.8%. The impact of mass flow rate and inlet temperature of water steam on condensation heat transfer coefficient is evaluated, but the impact is not significant.

(II) The final comparison of obtained results with other experimental studies shows maximal difference lower than 25%, more in Table 3. Correlation of results obtained from Wilson plot method with five tested equations for prediction of condensation heat transfer coefficient is 83.0%, 99.9%, 99.7%, 98.3%, 96.6% in order of equation Eqs. (11)–(15), more in Table 2. The first low correlation is obtained by Nusselt equation (Eq. (11)) without the wave's effect on condensation heat transfer coefficient.

(III) If the difference of results obtained from Wilson plot method and Nusselt equation is caused mainly by waves on condensate surface, then the wave's effect on condensation heat transfer coefficient is about 20.5% in this presented case.

Acknowledgment

This paper has been supported by research project LO1408 »AdMaS UP – Advanced Building Materials, Structures and Technologies« as research activity of AdMaS Centre.

References

[1] W. Nusselt, Die Oberflächenkondensation des Wasserdampfes, Z. Ver. Dtsch. Ing. 60 (27) (1916) 541–546.
[2] L.A. Bromley, Effect of heat capacity of condensate, Ind. Eng. Chem. 44 (12) (1952) 2966–2969.
[3] W.M. Rohsenow, Heat transfer and temperature distribution in laminar film condensation, Trans. Am. Soc. Mech. Eng. 78 (1956) 1645–1648.
[4] E.M. Sparrow, J.L. Gregg, Laminar free convection from a vertical plate with uniform surface heat flux, Trans. Am. Soc. Mech. Eng. 78 (1956) 435–440.
[5] E.M. Sparrow, J.L. Gregg, A boundary layer treatment of laminar film condensation, J. Heat Transfer, Trans. Am. Soc. Mech. Eng., Ser. C 81 (1959) 13.
[6] E.M. Sparrow, E. Marschall, Binary gravity-flow film condensation, J. Heat Transfer, Trans. Am. Soc. Mech. Eng., Ser. C (1969) 205.
[7] S.G. Bankoff, Stability of liquid flow down a heated inclined plane, Int. J. Heat Mass Transfer 14 (1971) 377.
[8] E. Marschall, C.Y. Lee, Stability of condensate flow down a vertical wall, Int. J. Heat Mass Transfer 95 (1973) 1.
[9] Q. Liang, Unsteady computational simulations and code developments for a study of internal film condensation flows stability, noise sensitivity and waviness (Ph.D. thesis), Technological University Michigan, 2003.
[10] Q. Liang, X. Wang, A. Narain, Effect of gravity, shear, and surface tension in internal condensing flows and results from direct computational simulations, J. Heat Transfer, Trans. Am. Soc. Mech. Eng. 126 (5) (2004) 676–686.
[11] A. Narain, Q. Liang, G. Yu, X. Wang, Direct computational simulations for internal condensing flows and results on attainability/stability of steady solutions, their intrinsic waviness and their noise sensitivity, Trans. Am. Soc. Mech. Eng. 71 (2004) 69–88.
[12] P.L. Kapitsa, Zh. Eksperim Teor. Fiz. 18 (1948) 3.
[13] W.H. McAdams, Heat Transmission, third ed., McGraw-Hill, New York, 1954.
[14] G.B. Whitham, Linear and Nonlinear Waves, Wiley, New York, 1974.
[15] T. Hobler, Heat Transfer and Heat Exchangers, Publishing House of Chemical Literature, Leningrad, 1961.
[16] A. Gollan, S. Sideman, On the wave characteristics of falling films, AIChE J. 15 (1969) 301.
[17] A. Miyara, Flow dynamics and heat transfer of wavy condensate film, J. Heat Transfer, Trans. Am. Soc. Mech. Eng. 123 (2001) 492–500.
[18] S. Thumm, Ch. Philipp, U. Gross, Film condensation of water in a vertical tube with countercurrent vapour flow, Int. J. Heat Mass Transfer 44 (2001) 4245–4256, http://dx.doi.org/10.1016/S0017-9310(01)00067-9.
[19] U. Gross, Th. Storch, Ch. Philipp, A. Doeg, Wave frequency of falling liquid films and the effect on reflux condensation in vertical tubes, Int. J. Multiphase Flow 35 (2009) 398–409, http://dx.doi.org/10.1016/j.ijmultiphaseflow.2009.01.001.
[20] H. Hausen, Wärmeübertragung im Gegenstrom und Kreuzstrom, Berlin, 1950.
[21] S. Kutateladze, Theory of Heat Transfer, Novosibirsk, 1970.
[22] A.F. Mills, R.A. Seban, The condensation coefficient of water, Int. J. Heat Mass Transfer 10 (1967) 1815–1827.
[23] R.A. Seban, J.A. Hodgson, Laminar film condensation in a tube with upward vapor flow, Int. J. Heat Mass Transfer 25 (1982) 1291–1300, http://dx.doi.org/10.1016/0017-9310(82)90123-5.
[24] S. Thumm, Ch. Philipp, U. Gross, Film condensation of water in a vertical tube with counter-current vapour flow, Int. J. Heat Mass Transfer 44 (2001) 4245–4256, http://dx.doi.org/10.1016/S0017-9310(01)00067-9.
[25] S.B. Al-Shammari, D.R. Webb, P. Heggs, Condensation of steam with and without the presence of non-condensable gases in a vertical tube, Desalination 169 (2004) 151–160, http://dx.doi.org/10.1016/j.desal.2003.11.006.
[26] I.S. Park, Numerical analysis for flow, heat and mass transfer in film flow along a vertical fluted tube, Int. J. Heat Mass Transfer 53 (2010) 309–319, http://dx.doi.org/10.1016/j.ijheatmasstransfer.2009.09.028.
[27] Z. Dengliang, X. Hong, S. Yan, Q. Baojin, Numerical heat transfer analysis of laminar film condensation on a vertical fluted tube, Appl. Therm. Eng. 30 (2010) 1159–1163, http://dx.doi.org/10.1016/j.applthermaleng.2010.01.031.
[28] S. Kim, Y.G. Lee, D.W. Jerng, Laminar film condensation of saturated vapor on an isothermal vertical cylinder, Int. J. Heat Mass Transfer 83 (2015) 545–551, http://dx.doi.org/10.1016/j.ijheatmasstransfer.2014.12.009.
[29] Z. Zhao, Y. Li, L. Wang, K. Zhu, F. Xie, Experimental study on film condensation of superheated vapour on a horizontal tube, Exp. Thermal Fluid Sci. 61 (2015) 153–162, http://dx.doi.org/10.1016/j.expthermflusci.2014.10.029.
[30] R.I. Pashkevich, P.V. Muratov, Film condensation in a large diameter tube with upward steam flow, Int. J. Heat Mass Transfer 81 (2015) 804–810, http://dx.doi.org/10.1016/j.ijheatmasstransfer.2014.11.001.
[31] F. Urban, M. Kubín, Ľ. Kučák, Experiments on the heat exchangers with the tubes of small diameters, AIP Conf. Proc. 1608 (1) (2014) 245–248, http://dx.doi.org/10.1063/1.4892743.
[32] X.-H. Ma, X.-D. Zhou, Z. Lan, Y.-M. Li, Y. Zhang, Condensation heat transfer enhancement in the presence of non-condensable gas using the interfacial effect of dropwise condensation, Int. J. Heat Mass Transfer 51 (2008) 1728–1737, http://dx.doi.org/10.1016/j.ijheatmasstransfer.2007.07.021.
[33] S.J. Kim, H.C. No, Turbulent film condensation of high pressure steam in a vertical tube, Int. J. Heat and Mass Transfer 43 (2000) 4031–4042 (PII: S0017-9310(00)00015-6).
[34] J.H. Goodykoontz, R.G. Dorsch, Local heat-transfer coefficients for condensation of steam in vertical downflow within a 5/8 inch diameter tube, National Aeronautics and Space Administration, Lewis Research Center (NASA), Technical Note (NASA TN D-3326), United States, Washington, D.C. 1966, [online: <http://ntrs.nasa.gov/archive/nasa/casi.ntrs.nasa.gov/19660008881.pdf>].
[35] H. Gröber, S. Erk, U. Grigull, Die Grundgesetze der Warme Übertragung, Springer Verlag, Berlin, 1963.
[36] J.W. Rose, Heat-transfer coefficients, Wilson plots and accuracy of thermal measurements, Exp. Therm. Fluid Sci. 28 (2004) 77–86, http://dx.doi.org/10.1016/S0894-1777(03)00025-6.